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Title: Development and Test of a new Concept for Biomass Producer Gass Engines

Abstract (max. 2000 char.): The technical requirements and the economical assessment of converting commercial diesel engine gen-sets into high compression spark ignition operation on biomass producer gas have been investigated. Assessments showed that for a 200 kW_e gen-set there would be a financial benefit of approximately 600.000 DKK corresponding to a reduction of 60% in investment costs compared to the price of a conventional gas engine gen-set.

Experimental investigations have been conducted on two identical small scale SI gas engine gen-sets operating on biomass producer gas from thermal gasification of wood. The engines were operated with two different compression ratios, one with the original compression ratio for natural gas operation 9.5:1, and the second with a compression ratio of 18.5:1 (converted diesel engine). It was shown that high compression ratio SI engine operation was possible when operating on this specific biomass producer gas. The results showed an increase in the electrical efficiency from 30% to 34% when the compression ratio was increased.

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Preface

The following transcript is the main report concluding the PSO-2005 project

Development and Test of a new Concept for Biomass Producer Gas Engines

(FORSKEL-6536) funded by Energinet.dk.

Appending the report are two preliminary reports.

1 Introduction

The purpose of the project was to develop and test a new SI-engine gen-set concept, custom-made for operation on biomass gasification producer gas. The concept is based on a commercial diesel engine gen-set. This involved the following tasks:

- A 20 kW_{el} diesel engine gen-set was rebuilt for high compression spark ignition (SI) operation on biomass producer gas.
- A special gas-train was developed for the engine. The gas-train incorporated an air/fuel mixing device that allows the load of the engine to be controlled by a gas blower which is included in the gasifier system.
- Experiments were conducted with two spark ignition gas engines, one with a compression ratio of 18.5:1 and one with a compression ratio of 9.5:1. Both engines have been operated with the same generator and the same test setup.
- Engine tests were conducted with the engines operating on gas form the Viking biomass gasification demonstration plant. Performance and emissions were investigated.
- An economical assessment has been made comparing the cost of converting a diesel- and a natural gas generator unit to run on biomass producer gas.

2 Engine Modification Requirements

The following section covers the rebuilding of a diesel engine, and a gas engine into producer gas operation. The design and construction of the gas train is also covered.

2.1 From Diesel to Producer Gas

When converting a diesel engine to run on producer gas, the engine has to be slightly altered. First, the diesel injection nozzles are replaced by spark plugs. The plugs are coupled to an ignition system, which controls the ignition timing and generates electricity to produce the spark. Second, a gas train must be installed, connecting the thermal gasifier and the engine. The gas train will cut of the gas supply in case of an engine stop. Third, air and fuel must be mixed before entering the engine. This is normally done by a carburettor, which assures a homogeneous mixture and makes it possible to regulate the air-fuel ratio (λ). A gas carburettor is coupled to a constant pressure regulator, ensuring that the pressure of the gas from the gasifier is constant.

However, the air-fuel ratio can also be controlled in another more simple way. In the case of the present study, the gas supply is controlled by a roots blower in the gasifier system, which also controls the productivity of the gasifier. When the generator coupled to the engine is connected to the electrical grid, the volumetric flow into the engine is constant, because the engine is operating at constant rpm, corresponding to the frequency of the grid. Then, if the gas flow is low, the engine will compensate by sucking in more air, because the volumetric flow is constant. In this way λ is changed. This method requires a mixing device with a minimum pressure loss, which also assures a homogenous mixture. Such a device can be seen in Figure 1. When controlling the performance of the engine in this way, expensive components like the constant pressure regulator and gas carburettor can be omitted.

Figure 1: Mixing device, Sulzer typ. 250 Y

An oxygen probe, which measures the oxygen content in the exhaust gasses, must also be attached to the engine. When operating on producer gas, the oxygen content in the exhaust gasses is an indirect indication of λ. The reason for this is that the composition of the producer gas varies, and the result is a shift in the heating value, the stoichiometrics, and the air consumption. For a precise control of λ, measurements of the gas composition must be made before the intake manifold, with subsequent adjustments of the roots blower. For commercial plants the oxygen probe is precise enough, and adjustments based on the gas composition are not necessary.

Also, a governor, controlling the throttle valve through an actuator and a magnet-pickup, needs to be installed. The governor control is only necessary during start-up, as the throttle valve will be fully open when operating on producer gas.

In a diesel engine the temperature of the air in the combustion chamber has to be high enough, for the diesel to ignite. Generally, the compression ratio in diesel engines is very high - between 14:1 and 25:1. Most diesel engines are built with a compression ratio around 17:1 - 18:1, which is higher than for a petrol- or natural gas engine. The general perception has been, that it is not possible to run an engine on producer gas at high compression ratios (higher than 14:1), without engine-knock. However, this has been proven wrong by former studies [1,2]. When converting a diesel engine to run on producer gas, the compression ratio should be left unchanged, since a high compression ratio improves the efficiency.

The gas train transports gas from the gasifier to the engine, and works as a safety feature in case a defect occurs on the engine. If the engine shuts down, valves in the gas train will immediately shut of the gas flow. This is controlled by a PLC (Programmable Logical Controller). The gas train for producer gas operation can be produced simpler than for natural gas operation. The main reason for this is that the gas from the natural gas grid is fed at an elevated pressure of e.g. four bars, while producer gas is fed at a pressure of little above atmospheric pressure. The engine and the gasifier are both connected to a common control system – the PLC, which shuts down both the engine and the gasifier in case of engine or gasifier failure. Due to this, the gas production and flow will stop in case of a shut down and thus the producer gas train only needs one shutter valve, whereas the gas train for natural gas needs two shutter valves for safety reasons.

The gas train for natural gas also needs a constant pressure regulator. This lowers the pressure of the natural gas to just above atmospheric pressure, so that air and gas can mix in a venturi or carburettor at constant air/fuel-ratio. A venturi/carburettor is not needed on a producer gas engine and because the air/fuel-ratio can be controlled by the gas blower via the oxygen probe in the exhaust system, the constant pressure regulator on the gas train in this operation is not needed either.

2.2 From Natural Gas to Producer Gas

A gas engine is running an otto-cycle, where air and fuel is mixed into a homogenous mixture, before it is sucked into the combustion chamber. Here it is compressed and ignited by a spark just before the top dead centre. If the fuel-air mixture is igniting spontaneously due to high temperatures, the engine will be exposed to a very high pressure which may damage it. This behaviour is referred to as *knocking*.

The natural gas engine works, in principle, in the same way as the producer gas engine, since there is no major difference in the combustion process of the two gasses. While the stoichiometric air/fuel-ratio is very low for producer gas e.g. 1.3:1, the stoichiometric air/fuel-ratio for natural gas is 11:1. The result is that the gas train needs to be dimensioned to allow a larger volume flow for producer gas than for natural gas. A natural gas engine normally has a compression ratio between 8:1 and 14:1. At higher compression ratios the engine will start knocking when operating on natural gas. Producer gas can, on the other hand, be combusted in engines with at high compression ratio without knocking. This implies that when rebuilding a natural gas engine, the compression ratio can be raised to improve the efficiency.

Raising the compression ratio can be done in several different ways. The most common way is to exchange the pistons with new and higher pistons, and thereby decrease the clearance volume. This approach is illustrated in Figure 2, and it is normally used when the cylinder head is flat and the combustion chamber is positioned in the top of the piston.

If the combustion chamber is in the cylinder head and the height of the pistons cannot be raised, a shaving can be milled of the bottom of the cylinder head. This will decrease the size of the combustion chamber and the compression ratio is increased, this approach is illustrated in Figure 3. Lowering of the cylinder head is not always possible, because of complications between parts in the engine. Nowadays, the engine production is optimized towards using as little material as possible, and on some engines it is therefore difficult to mill in the cylinder head without hitting cooling channels etc.

A third way to alter the compression ratio is by changing the thickness of the gasket between the cylinder head and the engine block. However, this is normally not a good alternative, since gaskets on modern engines are so thin that they can be difficult to alter effectively.

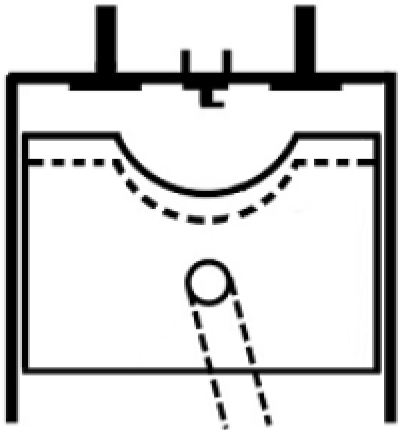

Figure 2: Clearance volume with combustion chamber in piston

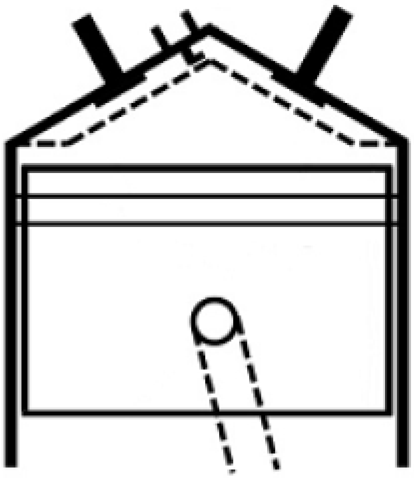

Figure 3: Clearance volume with combustion chamber in cylinder head

3 Economical Assessment

An economical assessment has been made comparing the cost of converting a diesel- and a natural gas generator unit to run on producer gas [3]. The frame of reference for the assessment is a generator unit with an output of 200 kW_{el}. The price difference for generator units with a higher output than 500 kW_{el} will be small, because diesel- generator units of this size are produced in smaller numbers. For engines with an output of 200 kW_{el} there is a large price difference because diesel engines of this size are produced in large numbers, compared to natural gas engines. The frame of reference of 200 kW_{el} is the final output of the converted engines. In the calculations, the difference in output before and after the conversion is taken into account. The cylinder volume of a diesel engine with an output of 200 kW_{el} is around 8-9 l, and around 12-14 l for a natural gas engine. To have the same frame of reference, the price of a natural gas engine with an output of 200 kW_{el}, is compared with the price of a diesel engine with a cylinder volume of 12-14 l. All prices are for 6-cylinder engines.

When natural gas was first introduced as an engine fuel, converted diesel engines were used to a large extent. The success rate of these engine conversions was low, and because of this the procedure was subsequently abandoned. As a result of this development, there is only minimal competence left in Denmark today, regarding conversion of diesel engines into gas operation. Therefore, it has been very difficult to gather quotes on this process, from other sources than the specialists at DTU. The overall assessment of the economics of the conversion process included; the time consumption of the specific conversion, the price of raising the compression ratio of a gas engine, and the total price of converting a diesel engine including spare parts.

It is estimated to require 60 working hours to convert a specific diesel engine model into operating on gas, and 20 working hours to convert any subsequent units of the same model. The price of one man hour is around 250 DKK/hour, and if the conversion is made by a company the costumer can expect to pay around 450 DKK/hour. This results in a conversion price of 27,000 DKK for a single conversion, and additionally 9,000 DKK for any subsequent conversions of the same model. These prices are only for the conversion, and not including the cost of materials. It is estimated that a total of 15 man hours are needed, when raising the compression of a gas engine. The price difference when producing multiple units is neglectable. The price of converting a gas engine, not including materials, is 7,000 DKK.

3.1 Rebuild of a Diesel Generator Unit

Numerous offers for diesel generator units have been collected. PM Energi A/S has provided prices on 4 different units. The units and their specifications are almost identical, and the price for the conversion is therefore almost the same. Because the price is almost the same, only one of the units is included in the assessment.

The different units considered all had the same cylinder volume (12-13 l), except the engines produced by the Chinese company Jinan Diesel on 35.7 l. The engines produced by Jinan Diesel are only operating at 1000 rpm where the rest of the engines are operating at 1500 rpm. The engines operating 1000 rpm will use a 6-pole generator and the engines operating at 1500 rpm will use a 4-pole generator - this produces an output frequency of 50 Hz. Because the engine from Jinan Diesel is only operating 1000 rpm, it is expected that the mechanical wear and friction will be lower, and the life time longer. Jinan Diesel uses the same engine block for producing 200 – 390 kW, with output depending mainly on the amount of turbo charging. Data on the efficiency of these engines is not available. Even though the Chinese produced engines differ a lot from the European ones, they are interesting because of the lower price.

When converting the engines it is necessary to install both an ignition system and a gas train. In addition, the diesel engine generator unit produced by Jinan Diesel has a compression ratio of 14:1 which is low compared to the other engines' compression ratios of 17.5:1. Therefore, the compression ratio for the Jinan Diesel engine has to be raised, to ensure optimal performance on gas. It is difficult to know the optimal compression ratio since the engine is operating at 1000 rpm. To reduce the complexity and price, it is possible to avoid the cascade regulation of the throttle valve, and thereby the governor, by installing a valve controlled by the DEIF controller as a positioner. A complete governor control costs about 13,000 DKK, while a positioner controller costs about 3,000 DKK – a cost reduction of 10,000 DKK. The price of the diesel generator units offered by PE Energi A/S, will be around 350,000-390,000 DKK after conversion.

A price comparison between the diesel generator unit from Jinan Diesel and PE Energi A/S can be seen in Table 1.

Table 1: Price of rebuilding diesel generator units [3]

Model	PM Energi A/S Volvo V410K	Jinan Diesel 250G
Generator unit	258,600 DKK	230,000 DKK/30,800 Euro
Ignition system	16,418 DKK	16,418 DKK
Simple Governor	3,000 DKK	3,000 DKK
Gas train	40,880 DKK + 9,706 DKK	40,880 DKK + 9,706 DKK
Oxygen probe	1,700 DKK	1,700 DKK
Mixing device	14,802 DKK	14,802 DKK
Rebuild of engine	27,000 DKK	6,750 DKK + 27,000 DKK
Total price	370,576 DKK	350,256 DKK

3.2 Rebuild of a Natural Gas Generator Unit

The natural gas generator unit produced by Jinan Diesel has the same specifications as the diesel generator unit, concerning the cylinder volume and the compression ratio. A comparative quote has been given by Nissen Energi Teknik which is a lot more expensive. The quote from Nissen Energi Teknik is concerning a gas generator unit including heat exchanger, natural gas train and control – and power board. The engine is a MAN E2876 LE302 gas engine with a compression ratio of 11:1. The price is in total 1,000,000 DKK. A quote for only the gas engine has been made by Frichs A/S costing only 380,000 DKK. The price difference between the gas generator unit and the gas engine by itself is 620,000 DKK. Because the compression ratio is too low, the engine needs to be rebuilt. Also a gas train for producer gas and a mixing device is needed. Furthermore an oxygen probe is needed to control the roots blower. Installed on the engine is an exhaust cooler which is not needed when operating on producer gas. On both the Jinan Diesel- and the Nissen Energi Teknik A/S a generator unit is installed with a governor. A price comparison between the gas generator unit from Jinan Diesel and Nissen Energi Teknik A/S is given in Table 2.

The price difference between the two quotes is 655,000 kr. The different configuration of the unit produced by Jinan Diesel does not necessarily pose a problem, but might actually be an advantage because of lower wear and friction. This makes the Jinan Diesel unit an interesting solution when operating on producer gas. Again it is noted, that the efficiency and the optimal compression ratio of the engine is not known. The quote by Jinan Diesel is especially cheap when compared to the other gas generator units, which generally costs the same as the unit produced by Nissen Energi Teknik A/S.

Table 2: Price of gas generator unit conversion [3]

Model	Jinan Diesel 250GF-TK	Nissen Energi Teknik A/S MAN E2876 LE302
Generator unit	345,000 DKK/46,200 Euro	1,000,000 DKK
Gas train	9,706 DKK	9,706 DKK
Oxygen probe	1,700 DKK	1,700 DKK
Mixing device	14,802 DKK	14,802 DKK
Rebuild of engine	6,750 DKK	6,750 DKK
Total price	377,958 DKK	1,032,958 DKK

Comparing Table 1 and Table 2 it is obvious, that a lot of money can be saved by converting a diesel generator unit compared to rebuilding a gas generator unit. Gas generator units are about 2.5-3 times more expensive than a converted diesel generator unit, and even the cheap gas generator unit made by Jinan Diesel is more expensive than the diesel generator unit.

Generally it seems like a significant reduction in investment can be archived by converting diesel generator units into producer gas units, but it should be noted, that the engine guarantees may be invalidated in the process.

4 Engine Test Setup

This section contains a description of the engine test setup at DTU. In the subsequent sections, the process and problems of the experiment are presented and discussed.

4.1 Experimental Setup

Production of Producer Gas

The producer gas, which the converted diesel engine generator unit operates on, is produced at DTU by the thermal gasification plant “Viking”. The Viking plant is a TwoStage down draft gasifier connected with an engine and generator. An overview of the system is presented in Figure 4.

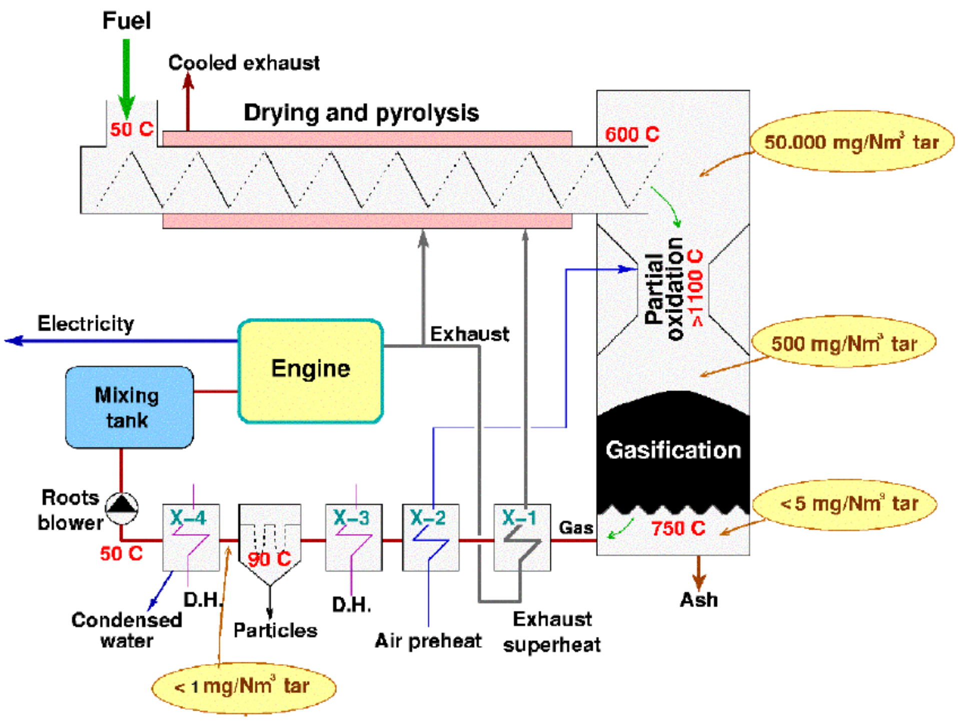

Figure 4: Thermal gasification plant at DTU.

The two-stage process is characterized by having pyrolysis and gasification in separate reactors with an intermediate high temperature tar cracking zone [4]. This allows for a very fine control of the process temperatures resulting in extremely low tar concentrations in the produced gas [5]. The Viking is a small-scale plant with a nominal thermal input of 75 kW. The plant is operating automatically and unmanned, it is fuelled by wood chips and the produced gas is fed to a gas engine.

The average composition of dry producer gas produced by Viking is:

$$33.3\ \%\ N_2,\ 30.5\ \%\ H_2,\ 19.6\ \%\ CO,\ 15.4\ \%\ CO_2,\ 1.2\ \%\ CH_4$$

The tar content is below 1 mg/Nm^3, the particle content is below 5 mg/Nm^3 and the average lower heating value (LHV) is 5.6 MJ/Nm^3.

The Engines

Experiments were conducted with two spark ignition gas engines of which one has a compression ratio of 18.5:1 and the other has a compression ratio of 9.5:1. Both engines have been operated with the same generator in the same test setup. The two engines are similar four-cylinder, four-stroke engines produced by Lister Petter. One is a factory-made natural gas engine with a compression ratio of 9.5:1 and the other is a custom made engine rebuilt from Diesel operation into producer gas operation with a compression ratio of 18.5:1. The cylinder volume of both is 1.86 litres [6].

The generator

Both engines were used in the same experimental setup alternately coupled to the same generator. The generator is produced by Leroy Somer and is a four-poled synchronous generator with an effect of 19.2 kW at cos(θ)=0.8. Because the generator is four-poled, it must run at 1500 rpm to be in sync with a 50 Hz grid.

The gas supply system

For the experimental setup there are two gas supplies to the operating engine: one for natural gas and one for producer gas. When the engine is operating on natural gas, the gas and air are mixed in a process where air is sucked in through the venturi, while natural gas is added through holes in the small section of the venturi. When operating on producer gas, air is sucked in through the venturi, while gas is added through a T-piece as shown in Figure 5.

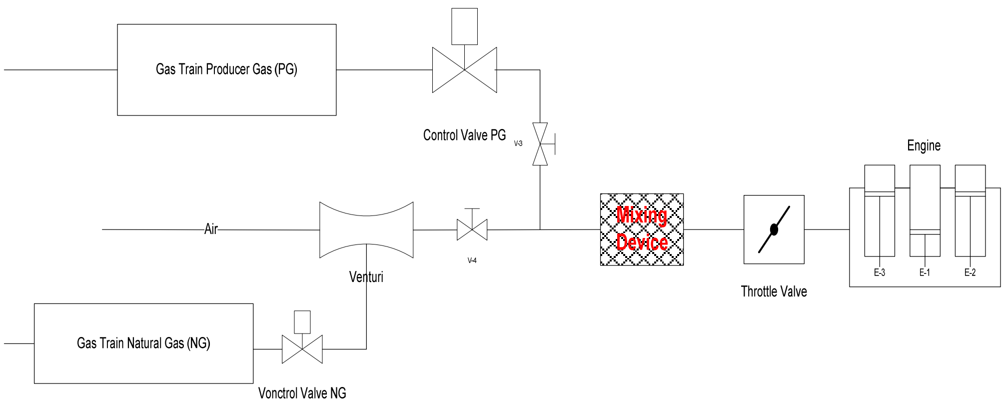

Figure 5: Gas supply system

To get a homogenous mixture, the air-gas mixture is lead through a mixing device in both cases. The venturi is used to suck in the air when operating on producer gas despite the fact that the pressure loss across a venturi is larger, compared to air just sucked in through a pipe opening. This is done instead of installing a new pair of valves for a separate air intake. When operating on producer gas this poses no problem, as the total pressure loss is low. The reason for this is that due to the lower stoichiometric air-fuel ratio of the producer gas, less air is sucked through the venturi in this process, compared to operating the engine on natural gas. The different gas trains can be seen in Figure 6.

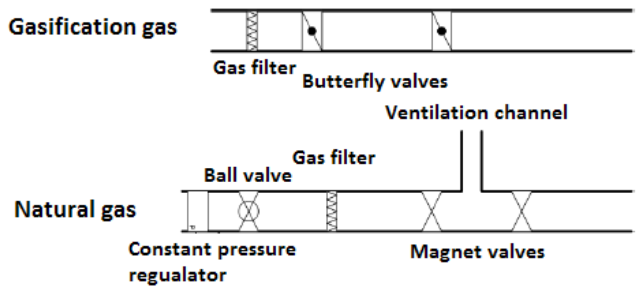

Figure 6: Gas trains

Natural gas is fed by an already established grid with a constant pressure of 4 bars. A constant pressure regulator reduces the pressure to just above atmospheric pressure. This makes the pressure difference between gas and air very small. Through the venturi a pressure difference is created, that sucks in natural gas. The pressure difference over the venturi is dependent on the volume flow, and therefore the air-gas ratio will be constant regardless of changes in the flow going into the engine. Positioned after the venturi a mixing device is placed that ensures a homogeneous mixture. When operating on natural gas the power of the engine is controlled by the throttle valve – this regulation form is termed quantitative regulation. See Figure 5.

The producer gas is fed directly from Viking, and air is supplied through the venturi. After this a mixing device ensures a homogenous mixture. When operating on producer gas, the throttle valve is electronically set to its fully open position. The power of the engine is now controlled by varying the amount of producer gas supplied. Because the generator is connected to the electrical grid, the grid will maintain a constant engine speed of 1500 rpm and the flow into the engine will therefore be constant. Due to this, the rest of the flow (air) will be sucked in through the venturi, and when the gas supply is varied the λ-value is varied accordingly. This regulation form is termed qualitative regulation.

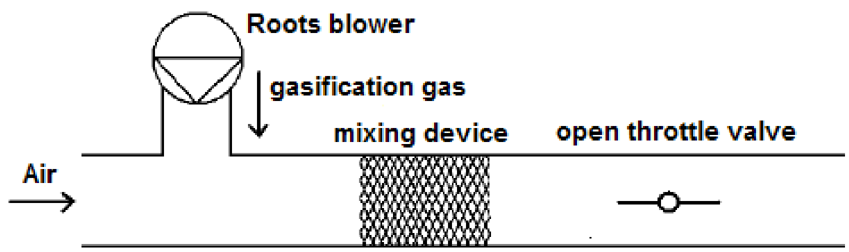

Figure 7: Simplified supply of producer gas

Electric Control System

In Figure 8 a component diagram of the electronic control is shown. The PLC is the highest ranked component in the system, because it controls the electrical grid connection and checks if various safety parameters are complied with. In case the parameters are not complied with, the PLC will shut down the DEIF controller and the engine stops. The job of the PLC is to ensure that the engine does not make the existing electrical grid unstable. The signals are sent continuously, but they are described as a series of events to simplify the description. The DEIF controller is the second highest ranked control unit. This unit controls the synchronisation of the generator and the electrical grid. During start-up the DEIF controller receives the actual frequency signal from the generator and compares this to the 50 Hz, which the DEIF controller is set to maintain. The DEIF controller then sends a signal ("set-point") through the integral band to the governor, which tells how fast the DEIF controller wants the engine to run. At the same time, the governor receives a signal from the engine about the actual rotational speed of the engine, measured by a magnet pickup. On the experimental setup these signals are between 4-20 mA. The governor compares the two signals, and sends a signal to the actuator about how much, it should open or close the throttle valve. When the engine is operating at exactly 1500 rpm the generator is coupled to the grid and the grid will maintain a constant engine speed of 1500 rpm. When the generator is operating parallel with the grid the DEIF is not controlling the governor anymore and therefore the throttle valve position can be controlled by a set-point from the PLC.

When operating on producer gas, the engine performance is controlled by controlling the roots blower instead of controlling the throttle valve. The performance of the engine will be dependent on the signal that was sent to the actuator or roots blower. As the engine is directly connected to the generator, this will give a new signal to the DEIF controller, and the cycle starts all over. After start-up, it only takes a short while, before a steady engine performance is achieved.

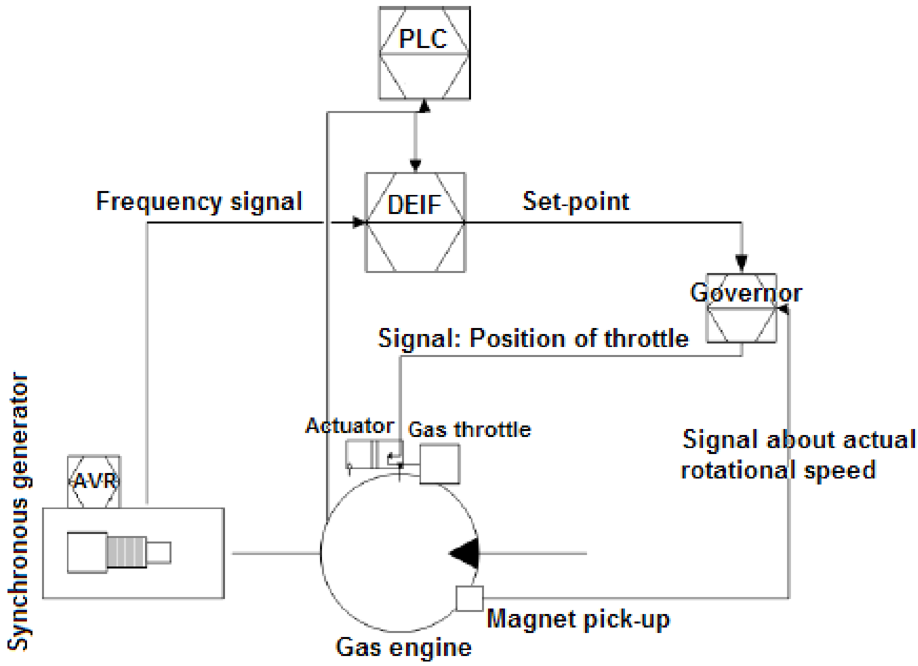

Figure 8: Electronic control of engine

Shift from Natural Gas to Producer Gas Operation

The system is started up, operating the engine on natural gas. When production of producer gas becomes stable, the shift from natural gas operation to producer gas operation can be commenced. This is done by slowly opening the producer gas supply. This will increase power output above the set-point and the PLC will then begin to reduce the natural gas flow. When the control valve for the natural gas flow is closed, the PLC will shift to producer gas operation mode and the throttle valve will shift to full open position. The performance of the engine is now controlled by the producer gas flow.

Cooling system

The cooling system is shown in Figure 9. A part of the cooling system acts as a fictive central heating system and this makes it possible to make calculations of the efficiency of the whole system. The central heating system and the engines cooling system are connected through a heat exchanger, and the heat exchanger is connected to a 3-way valve and a thermometer. The temperature of the cooling fluid going into the engine is measured, and if this is below set-point, the valve guides the cooling fluid (glycol) in the external central heating system around the heat exchanger. If the temperature is high, the cooling fluid will run through the heat exchanger, and thereby cool the engine.

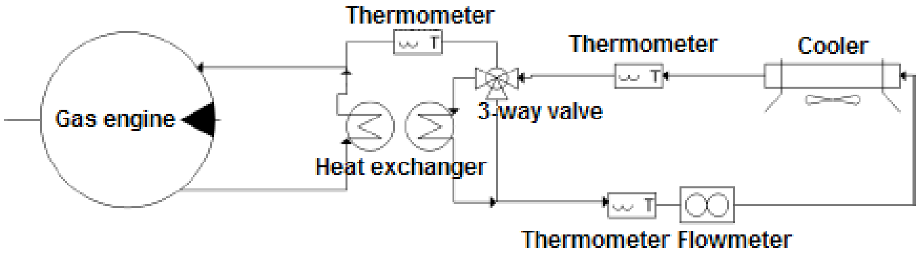

Figure 9: Cooling system

The Experimental Setup

Figure 10 shows the experimental setup including the different components and the measuring points for data collection.

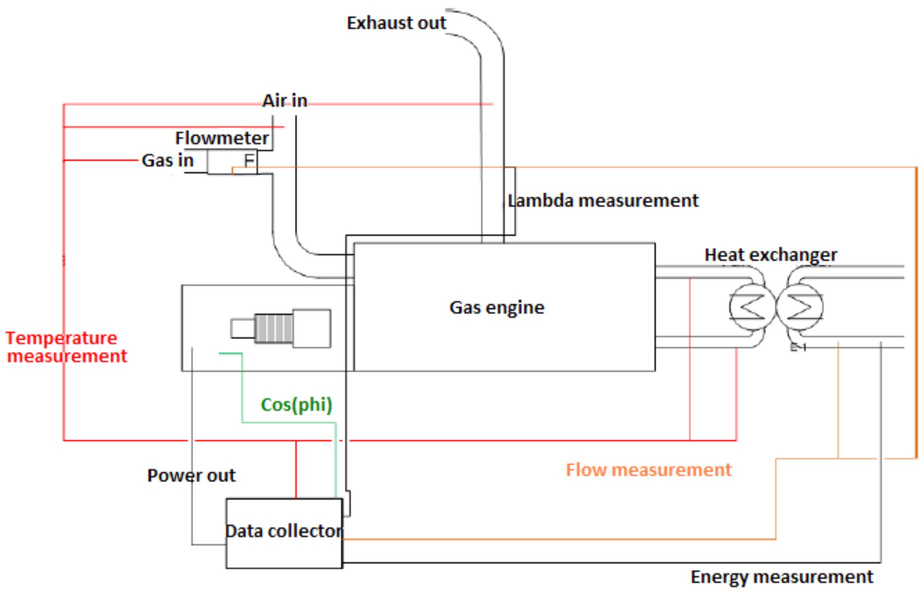

Figure 10: Experimental setup

4.2 Conducting Experiments

As mentioned before, the engine will start on natural gas until a steady engine performance and connection to the electrical grid have been achieved. When the engine is connected to the electrical grid, the fuel is changed from natural gas to producer gas as described in the section above. After another steady engine performance has been achieved, data can be collected. Temperature measurements are made on the inlet- and outlet of engine cooling fluids and in the heating system the temperature and flow is measured. The temperature and the O_2 content (measured through an oxygen probe) of the exhaust gasses are measured, and the temperature and flow of the producer gas are measured before entering the engine. The electrical output is also measured.

4.3 Experimental Trouble-Shooting

During the first operation of the test setup, several problems were encountered. The problems concerned the regulation of -and the gas supply to, the engine. When the experiment was first initiated, it was not even possible to run the engine on producer gas, and only short time operation on natural gas was achieved.

One of the most time consuming problems was to get the gas flow stable. A valve was stuck, which meant the valve was not open, even though it appeared so. Because the defect was not visible, the venturi was thought to be the problem. The dimensions of the venturi were changed before the real problem was located. It was thought that the air-fuel ratio was too high, and more holes were drilled in the venturi. After the problem with the valve was discovered, the air-fuel ratio became too low, and 1/3 of the holes were sealed with tape. This changed the air-fuel ratio to a satisfying value.

Another problem was that the engine had difficulties operating at low rpm. After a time consuming error detection process, it was found that the constant pressure regulator did not reduce the pressure of the natural atmospheric pressure. This meant that the gas was pressed out through the holes in the venturi, instead of being sucked out due to the pressure difference over the venturi. The settings of the constant pressure regulator were changed and now the pressure was slightly above atmospheric pressure. The problems with the engine were solved – the gas supply was constant, and it was stable as soon as the right air-fuel ratio was found.

Problems with the DEIF and the Governor

There have been numerous problems with the programming of the DEIF controller. The whole system including the DEIF controller is very complex.

It has been a problem to get the DEIF and governor to communicate properly. The DEIF sends out a signal of 4-20 mA. This signal is then transformed into a signal between 0-10 V before going to the governor. The problem was in the transformer, where the input impedance was too high, which meant that the transformer was unable to hold the voltage. Because of this the signal was not forwarded to the governor as a signal between 0-10 V. The problem was solved by changing the transformer to one with an external power supply.

There was also a problem with the governor, which controls the throttle valve through an actuator. The control was very unstable. It turned out that the problem was the time constants within the two regulations in the cascade regulation, which were too close to each other. This meant that the governor did not have enough time to adjust itself to the set-point from the DEIF, before it got a new set-point from the DEIF. The settings of the proportional- and integral band in the DEIF were changed, and the amplification of the signal was lowered. This was done in small steps until stability was achieved.

After this the generator was successfully connected to the electrical grid, but it quickly dropped off the grid again. After having contacted the manufacturer of the DEIF controller, it turned out that the leads sending feedback to the DEIF was placed wrong. This meant that the system registered a negative power production from the generator.

Another problem was the error message “high voltage”, which forced the system to shut down. The problem lay within the automatic voltage regulator – the AVR. The error originated from a misplaced component in the AVR. Correction of this error was difficult and time consuming, as the generator had to be partly disassembled, every time an adjustment was to be made.

Signal error

When the experiment was first initiated, it was only possible to start the engine once a day. If it was attempted to start up the engine again, the governor regulator did not work, and neither did a series of alarms that was supposed to stop the engine.

At first, the problem was believed to be caused by the offset, which turned the DEIF regulation on, after the PLC had released the control of the engine to this control unit. The offset was set to 0 seconds, because the DEIF must take over control immediately after the PLC releases the engine (over 400 rpm). It was assumed that the problem lay within the generator, as a small amount of voltage was still left in the generator from the first startup. Because the governor is regulated by a signal from the DEIF controller, as well as from a signal from the generator, it is assumed, that the voltage left in the generator disturbed the signal. This disturbance of the signal made it difficult for the governor to make any regulations. The solution to this problem was to set the offset to 8 seconds. This gave the voltage left in the generator time to escape from the system, and resulted in fewer problems during multiple startups. However, this solution did not solve everything completely as there were still some startup problems.

5 Modeling

To simulate the energy flows through the engine a one-zone model was constructed. This model gives theoretical data about how the engine will work under different conditions, and these data can then be compared to real measured data. The values computed in the program are not expected to be consistent with reality, but the tendency in the data is [7].

A one-zone model simulates an operating engine at different parameter variations. There are different factors that are not taken into account in the model, and these will be mentioned in connection with the relevant diagrams later on in this section.

5.1 The Compression Ratio

The compression ratio of an engine describes the relationship between the volume in the cylinder when the piston is in the bottom dead centre, and when the piston is in the top dead centre position – that is how much the air-gas mixture is compressed. When the piston is working its way up the cylinder, the mixture is compressed and both temperature and pressure is rising. If the pressure and temperature becomes too high the air-gas mixture can self-ignite. This means that the air-gas mixture ignites and expands very rapidly, while the piston is still working its way up through the cylinder. Normally the air-gas mixture would be ignited by a spark plug before the piston reaches the top dead centre, which gives a controlled burn of the fuel. When the air-fuel mixture self-ignites the pressure and temperature increases uncontrolled, like an explosion, and this can damage the engine. This is called knocking. A typical natural gas engine has a compression ratio around 12:1 and if the compression ratio is higher, the tendency towards knocking will increase.

In an ideal Otto cycle it is assumed that the working fluid is an ideal gas and that compression and expansion is adiabatic, in this case the indicated fuel conversion efficiency is equal to [8]:

$$\eta_f = 1 - \frac{1}{CR^{\gamma-1}} \qquad (1)$$

where:

CR *is the compression ratio*

$\gamma = \frac{C_p}{C_v}$ *is the ratio of specific heats*

C_p *is the specific heat at constant pressure*

C_v *is the specific heat at constant volume*

From the equation it is evident that the indicated fuel conversion efficiency increases with increasing compression ratio and decreases as γ decreases. The latter is due to a reduction in exhaust gas temperature.

Increasing compression ratio is the most significant way to increase the thermal efficiency. However, the benefits of increasing the compression ratio are limited by a tendency towards engine knocking and additional undesired process emissions.

Experimental work has shown that the compression ratio is normally limited by the octane rating of the fuel. For gaseous fuels, the methane number is used – instead of the octane rating – to compare knocking tendencies for different fuels. Methane has a high octane rating and this rating is raised further by the presence of CO_2 in the gas. On the other hand, the presence of hydrogen in the gas is expected to cause problems with knocking. Danish natural gas has a methane number of approximately 70 [9] and the producer gas is expected to have a methane number of approximately 50 [10].

It has also been shown that both efficiency and MEP (Mean Effective Pressure) have maximum values around a CR of 17 depending on other engine parameters [8]. This trend is due to an increasing surface/volume ratio as the CR increases, likewise the effect of crevice volumes increases and the combustion becomes slower [8].

As mentioned above, the compression ratio applicable for the available gaseous fuels is limited by the methane number and the maximum for natural gas engines is nowadays around 12.

When operating an engine on producer gas the tendency towards knocking is smaller, because of the high content of inert gasses and the faster flame speed due to the higher content of hydrogen. Compression ratios of up to 17:1 have shown good results, without any sign of knocking [11]. The engine used in the present experiments has a compression ratio of 18.5:1.

When the compression ratio is raised, the efficiency of the engine will increase, and the temperature of the exhaust gasses will fall. This happens because the gas expands more before it reaches the exhaust and also because the combustion efficiency is higher. One consequence of this is that less energy is available in the exhaust gas, which could be used to run a turbo. As the temperature of the exhaust gasses decrease, the density increase. This results in an equivalent mass flow but a smaller volumetric flow and therefore, less work is delivered to run a turbo.

When the compression ratio is raised, more strain is put on the cooling system since the extra amount of produced energy also produces more heat. To run an engine at high compression ratios require that the engine is sufficiently strong. The stress on the crankshaft and connection rods becomes larger, and these have to be dimensioned to handle the larger stress.

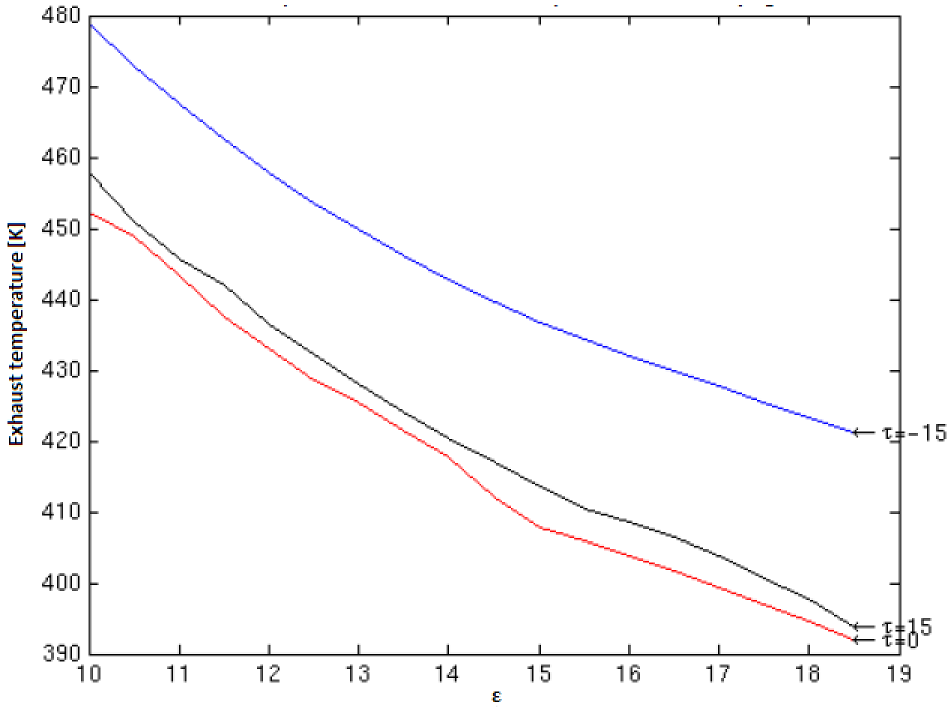

Figure 11: Exhaust temperature as a function of compression ratio (ε) at varying ignition timing (τ, deg. crank shaft angle before top dead centre).

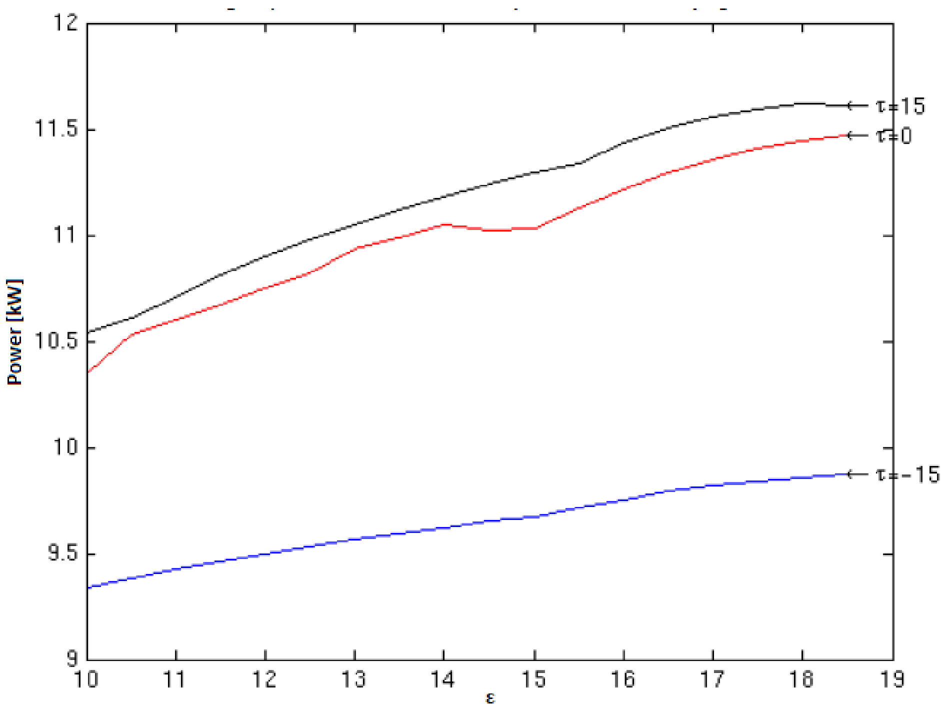

Figure 12: Engine power as a function of compression ratio (ε) at varying ignition timing (τ, deg. crank shaft angle before top dead centre).

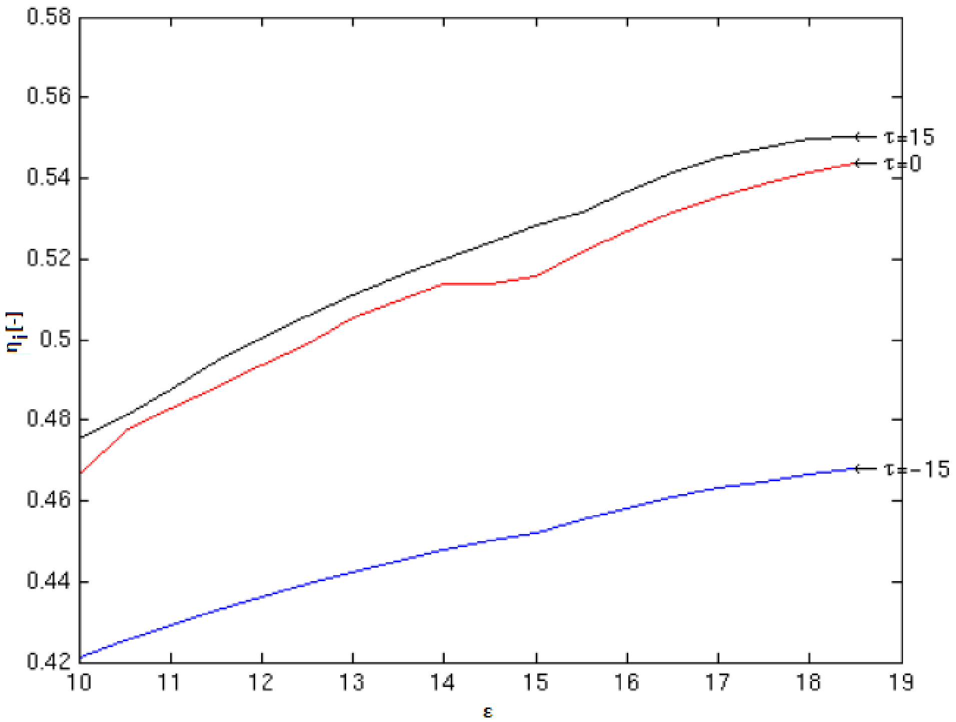

Figure 13: Engine efficiency as a function of compression ratio (ε) at varying ignition timing (τ, deg. crank shaft angle before top dead centre).

5.2 Analysis of Theoretical Data

In Figure 11 - Figure 13 the results from the one-zone model are shown. As seen on Figure 11, the exhaust temperature is dropping when the compression ratio (ε) is raised. This is consistent with the theory saying that more energy is generated inside the engine. This effect can also be seen on Figure 12 which shows engine power. The power is rising as the compression ratio is increased, and this also agrees with the theory. On Figure 13 the engine efficiency is shown. This value is also increasing.

Engine Power as a Function of Compression Ratio and Ignition Timing

On Figure 12 the engine power is shown as a function of the compression ratio at varying ignition timing settings. Generally, it shows an increasing tendency, but the graph is leveling off at high compression ratios. This means, that at a given bore and length of stroke, a point can be reached where a further compression ratio enlargement, does not increase the power produced by the engine. The ignition timing is varied at the same time, with the following values $\tau = \{-15^o, 0^o, 15^o\}$. In the model, the ignition delay is set to zero and 100 % combustion is assumed to take 30 degrees. The power produced increases, the closer the ignition point is set to the top dead centre. However the power is increasing at $\tau = 15^o$ – an ignition at 15^o after top dead centre. This is because the model does not take into account the varying combustion duration according to ignition timing.

If the ignition timing is set before top dead centre ($\tau = -15^o$), the pressure inside the cylinder will rise while the piston is still moving towards the top dead centre. The build up pressure will then work against the piston. The chemical energy being transformed when the gas is burned can be converted in two ways, either as heat or as kinetic energy. Because the movement of the piston is opposite the expansion of the gasses inside the cylinder, the energy is converted into heat. Thereby, the heat production inside the cylinder increases, while the mechanical energy decreases. At the same time the surface (where heat conduction is possible) is larger (cylinder wall + cylinder head + piston) because the piston has not reached the top dead centre. Therefore it is easier for the produced energy to 'escape' from the cylinder. Because of the big loss of energy to heat production, the curve for an ignition timing of -15^{o} is positioned below the two other curves.

When the pressure is build up, the temperature is rising which makes the combustion time shorter – this is not taken into account in the model. If the ignition timing is set after top dead centre (τ = 15^{o}) the combustion time will be longer than the 30 crankshaft degrees used in the model. Thereby the power production is diminished, as the pressure and temperature in the cylinder is decreased. If the temperature in the cylinder becomes too low, the flame front will die. This means that there will be a small amount of gas left in the exhaust, which never got combusted and converted into mechanical energy. These factors are not taken into account in the model, and because of this the graphs may only show a tendency – that the power produced is increasing as the compression ratio is increasing, and that this is the case for any ignition timing.

At high compression ratios, the combustion of the gasses will be slower [8]. If the compression is low, a large space will be available where the flame front can spread as a hemisphere shape. If the compression is high, the flame front will quickly reach the top of the piston and be stopped. After this the flame front will only develop in a radial direction, and thereby burn slower than at lower compression. This is shown in Figure 14.

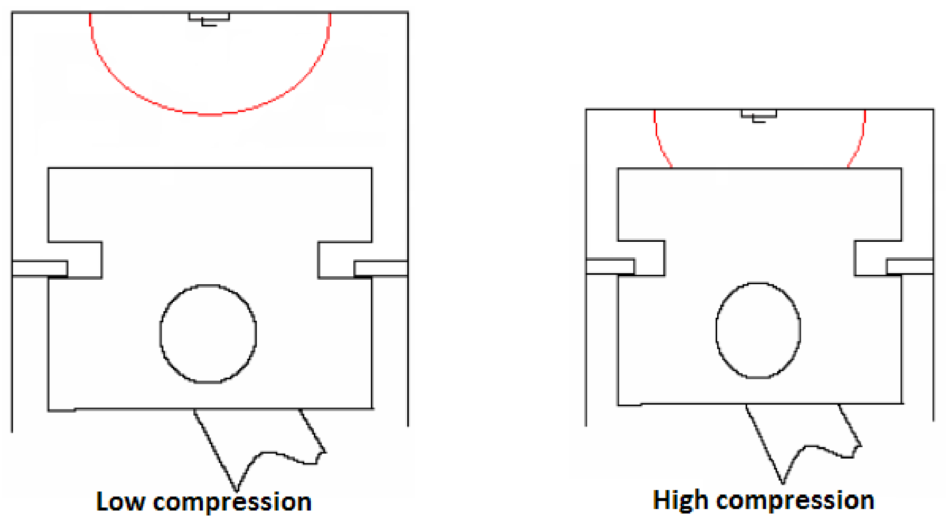

Figure 14: Low and high compression

Engine Efficiency as a Function of Compression Ratio and Ignition Timing

The engine efficiency is shown in Figure 13, and the tendency depicted is the same as for the engine power. The curve for the engine efficiency will level off, where a further increase of the compression ratio do not cause any significant change in the efficiency. This makes good sense since the efficiency is calculated on the basis of power produced and the input energy, and the ratio between these are increased when the compression ratio is raised (as shown in Figure 12 the power is increasing until a certain point, even though the same amount of air-gas is combusted). The efficiencies are relatively high – at a compression ratio above 12, the engine efficiency for $\tau = 0^o$ is more than 50 %. This is very high, but again the graphs must be interpreted qualitatively and not quantitatively. The efficiency of a real engine will be lower, even though the specifications of the engine might be the same as the simulated. Again, if the ignition timing is 15 degrees before top dead centre ($\tau = -15^o$), the efficiency is significantly lower than for an ignition timing of 0^o or 15^o. As already mentioned, this is because the pressure build up in the cylinder is working against the piston in the model. The curve for $\tau = 15^o$ is a theoretical value because the combustion time is not altered under changing temperature- and pressure conditions. The temperature might become so low, that a complete combustion is not possible.

Exhaust Temperature as a Function of Compression Ratio and Ignition Timing

Figure 11 shows that the exhaust temperature is dropping as the compression ratio is rising. One reason for the decreasing exhaust temperature is that the combustion is happening in a smaller volume, while the surface area/volume ratio is increasing. Therefore, larger heat conduction is possible when compared to the combustion volume. Another reason is that the increased compression ratio will create a larger pressure during combustion. When the pressure is rising the temperature rises as well. Large heat conduction will then occur and afterwards an expansion to atmospheric pressure takes place. The expansion lowers the temperature even more, and this effect will be more significant at higher compression ratios.

6 Results

The following section describes the results of the experimental work in the project.

Experiments were conducted with two identical Lister Petter engines, one with a compression ratio of 18.5:1 and one with a compression ratio of 9.5:1. Both engines have been operated with the same generator in the same test set-up as described above. The experimental work included measurements of performance and emissions while varying both the fuel/air ratio (λ) and the ignition timing. Additional experiments were made with turbo charging of the high compression engine. All experiments were conducted with full open throttle, and the engine load was controlled by varying λ.

The gas composition varied considerably during the experiments due to significant variations in fuel moisture. Figure 15 shows examples of the gas composition from tree different experiments.

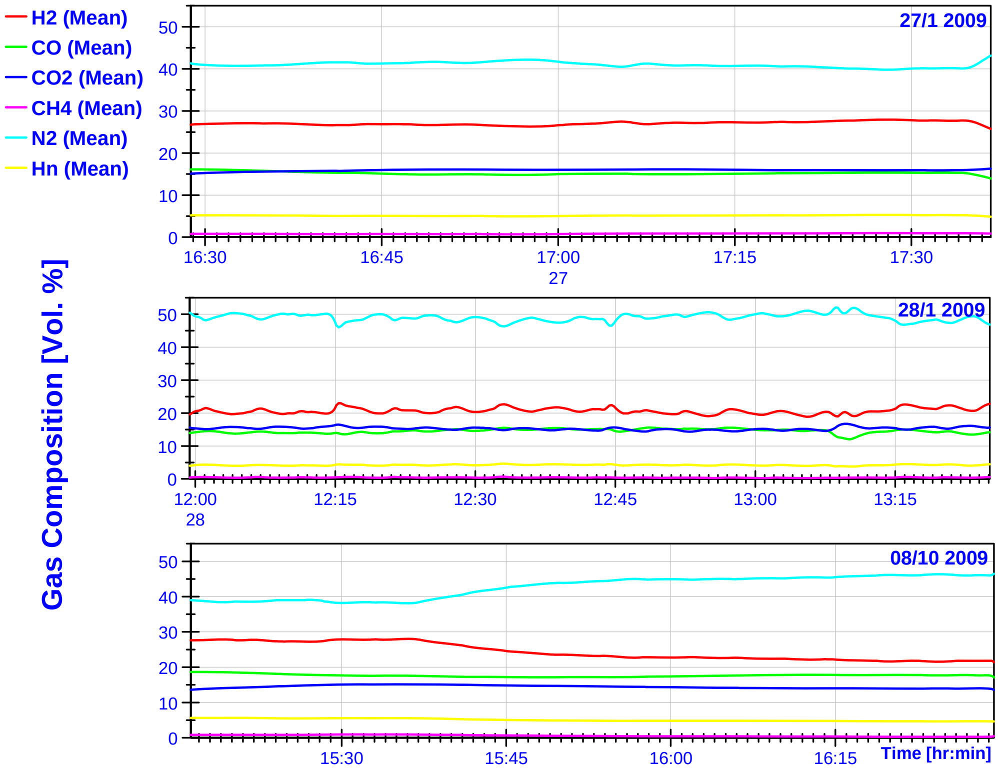

Figure 15 Examples of gas composition measurements

6.1 Performance

Figure 16 shows the performance of the high compression engine when varying λ and thus the load. This is comparable with the results in Figure 17 for the low compression engine. It is evident that there is a significant increase in efficiency when going from low compression to high. This is especially distinct at high load (λ=1.4) were the efficiency increases from 30% to 34%. This results in an increase in power out from 8.9 kW to 10.1 kW corresponding to an increase of 13.5%.

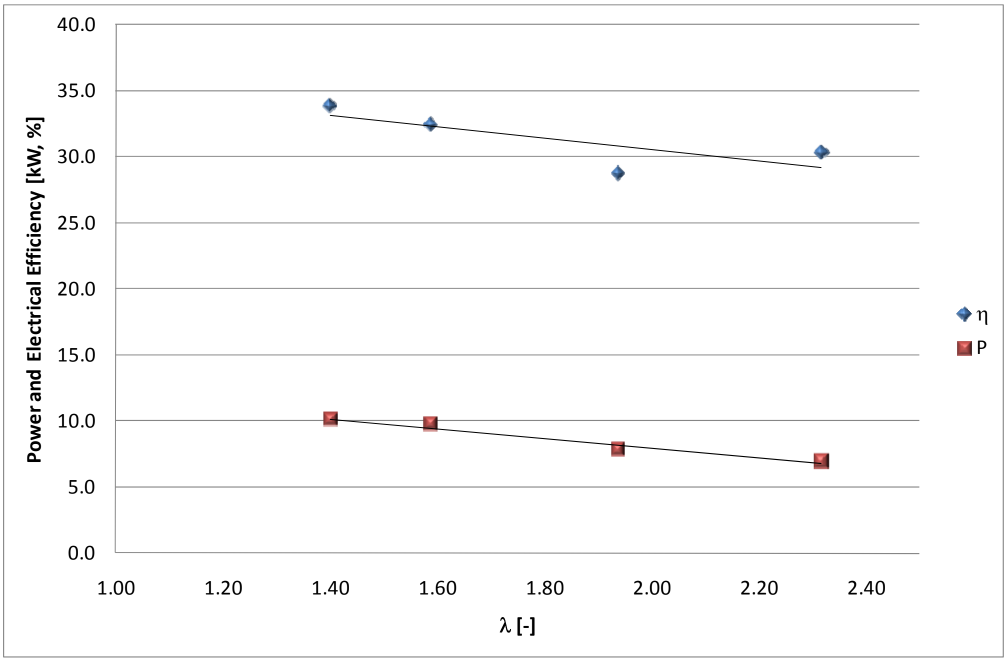

Figure 16 Power and electrical efficiency depicted as a function of λ, CR=18.5:1.

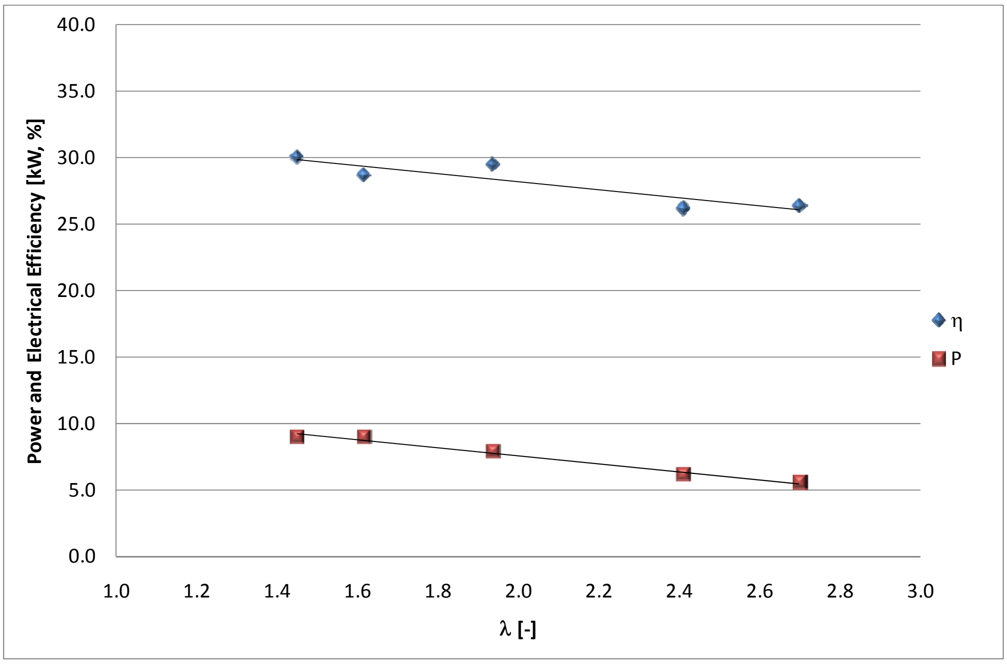

Figure 17 Power and electrical efficiency depicted as a function of λ, CR=9.5:1.

Figure 18 shows the relationship between the performance and the ignition timing in different load situations for the high compression engine. It is seen that the trend for λ=1.4 and 1.6 is a slight decrease in the power output while for λ=1.9 and 2.2 no influence of advancing the ignition timing is apparent. A decreasing trend in the efficiency is registered for all λ-values, when the ignition timing is advanced.

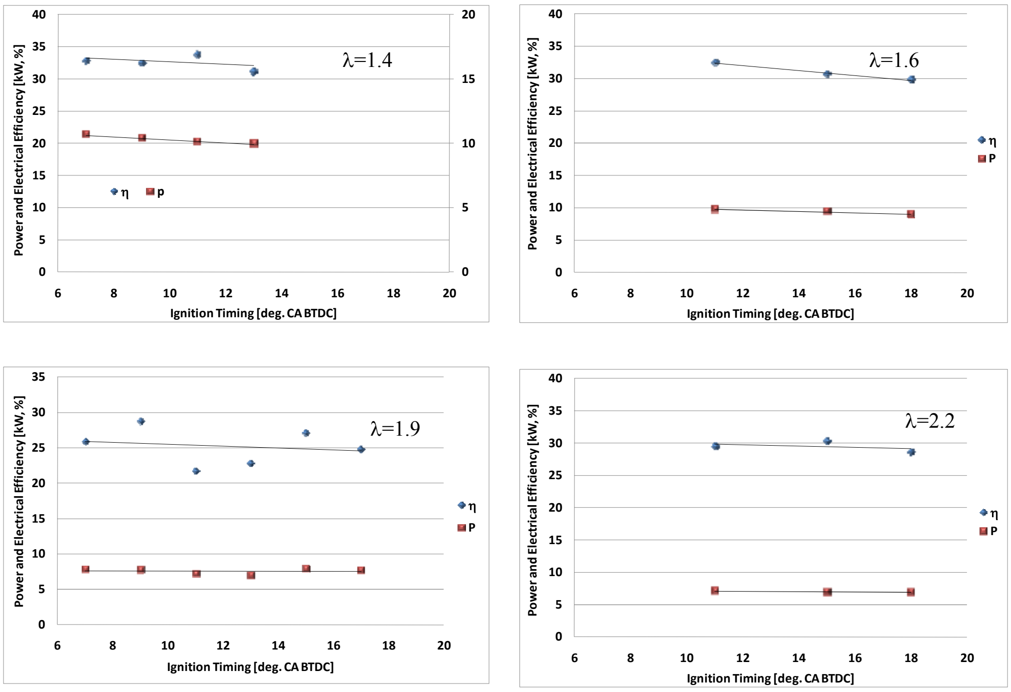

Figure 18 Power and electrical efficiency as function of ignition timing, CR=18.5:1

Ignition timing has a significant effect on cylinder peak pressure and temperatures and thus on the NO_x emissions [8]. Retarding the ignition timing reduces NO_x emissions and it is therefore interesting that maximum power and efficiency values are seen for retarded ignition timings. In section 6.2 the results of the corresponding emission measurements are presented. It is shown how the NO_x emission may be reduced significantly by retarding the ignition timing.

Figure 19 shows the relationship between the performance and the ignition timing for the low compression engine at λ=1.4. In this case, opposite the high compression engine, no influence on efficiency is registered as a function of advancing the ignition timing. The same behavior is seen for the power output.

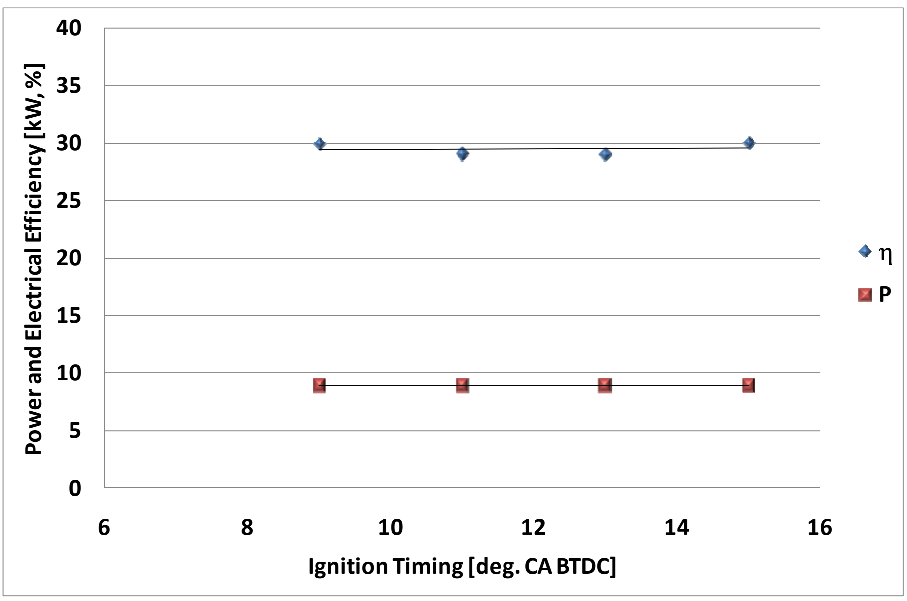

Figure 19 Power and electrical efficiency as function of ignition timing, λ=1.4, CR=9.5:1

6.2 Emissions

Figure 20 shows the NO_x and CO emissions from the experiments with the high compression engine. The values depicted are for optimum efficiency operation at each load situation. The trend, that CO is increasing while NO_x is decreasing with increasing λ is similar to what is usually seen for producer gas engines [1,11]. The magnitude of the emissions is also similar to what has been experienced from other engines operating on gas produced by the Viking gasifier.

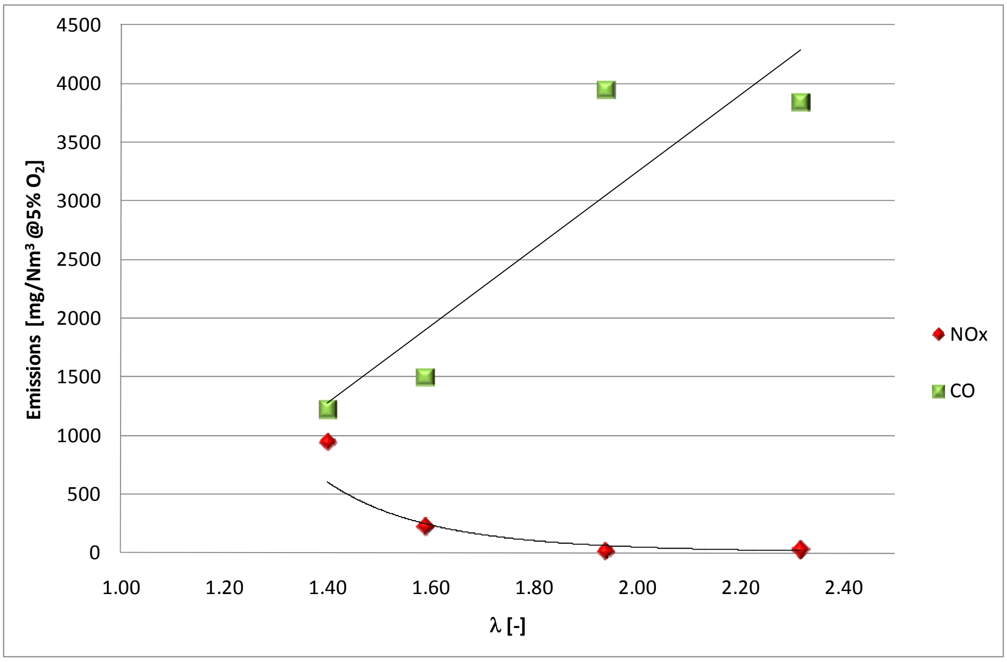

Figure 20 NO_x and CO emissions depicted as a function of λ.

When looking at the dependency of the emissions with regards to ignition timing, the NO_x emission is especially interesting. As is seen in Figure 21, there is a significant decrease in the NO_x emission when the ignition timing is retarded. This trend is predominating at low λ values, and this is similar - but much more distinct, to what is seen for a low compression engines operating on gas form the Viking gasifier [11]. For λ=1.4 emission decreases by almost a factor 3 when the timing is retarded from 13^o to 7^o before top dead center. The measurements indicate that it is possible to operate the engine at λ=1.4 and comply with the emission control limit for NO_x which is 550 mg/Nm^3. Retarding the timing reduces the NO_x emission because the peak pressure during the combustion is decreased as more fuel is burned after top dead center [8]. For lean operation (high λ values) virtually no NO_x emission is seen.

The trend for the CO emission shows increasing emission when the ignition timing is advanced, which is opposite to what is seen for a low compression engine operating on gas from the Viking gasifier [11]. The magnitudes of the CO emissions are higher for the high compression engine than what is seen for low compression engines [11], and particularly for lean operation there is a significant difference. This is caused by an increase in unburned fuel emission. Because of the high compression ratio relatively more fuel will be trapped in crevices and the post combustion will be less efficient due to a lower exhaust temperature [12].

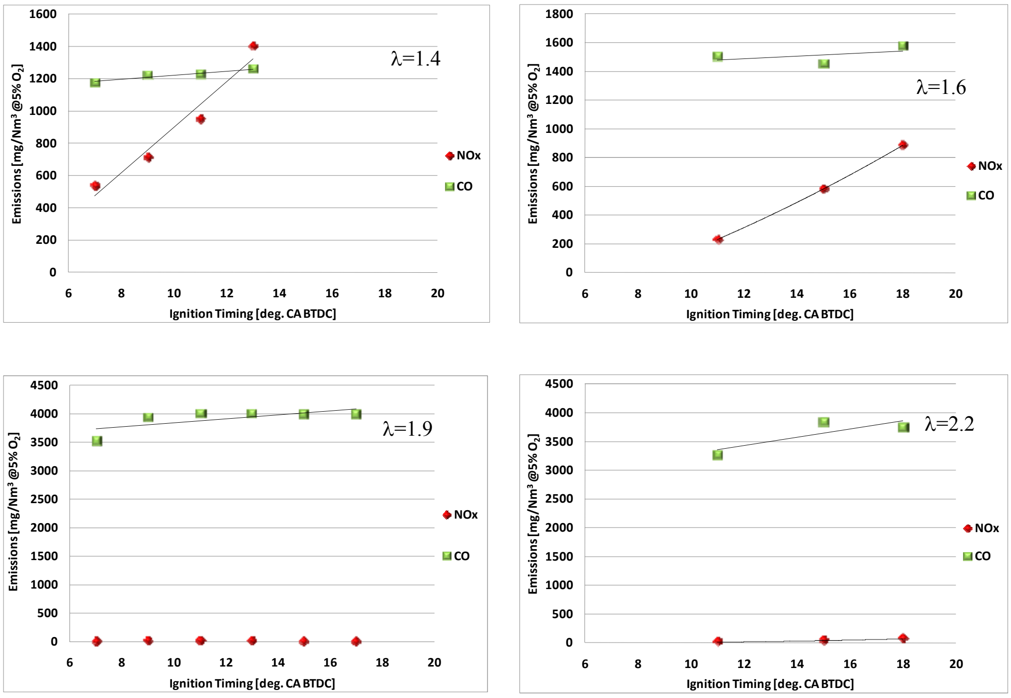

Figure 21 NO_x and CO emissions depicted as function of ignition timing, CR=18.5:1

6.3 Turbo Charging

Experiments have been conducted with low pressure turbo charging of the high compression engine. The aim has been to initiate investigations regarding high compression pressurized engines operating on producer gas. The turbo charger used for the experiment originated from a diesel version of the engine which is designed and optimized for stoichiometric engine operation on diesel. This means that operating the turbo on low temperature exhaust gas will result in only a week pressurizing of the air/fuel charge.

The intake pressure was increased from approximately 500 mmWG under-pressure to approximately 600 mmWG over-pressure. The low pressure charging resulted in a 10 % increase in power output compared to the uncharged high compression engine and 20 % compared to the low compression engine - see Figure 25. The efficiency was apparently slightly diminished due to the pressure charging for low λ-values - see Figure 26. This may be due to an increase in back-pressure in the exhaust system which was not optimized for pressure charging. Also for the low λ measurements with pressure charging, the gasifier system was struggling to deliver sufficient amounts of gas resulting in slightly unstable engine operation - see Figure 22. This is also reflected in the measurements shown in Figure 23, where it is seen that the efficiency apparently increase slightly when λ increases. The trend for the power output is similar to what is seen for uncharged engine experiments.

The emission measurements show trends similar to what is seen for the uncharged experiments but it appears that the emission level is slightly increased. However, the NO_x measurements are quite scattered, and this may once again be due to the slightly unstable engine operation described above.

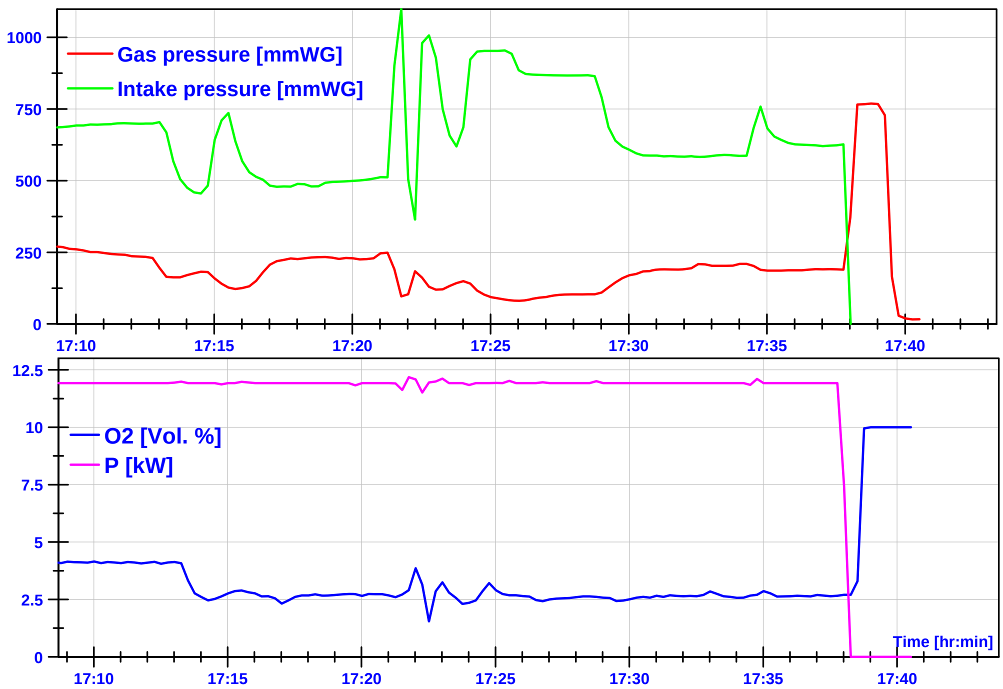

Figure 22 Measurements of gas pressure, intake pressure, exhaust O_2 and power during the high load turbo charged experiments.

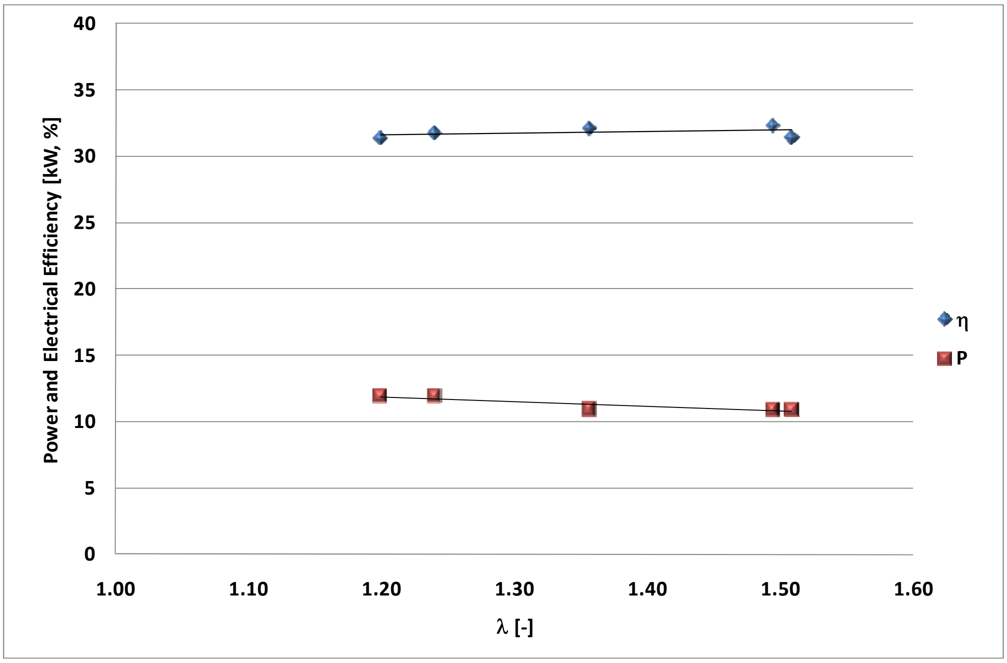

Figure 23 Power and electrical efficiency as function of λ, 18.5:1 turbo charging

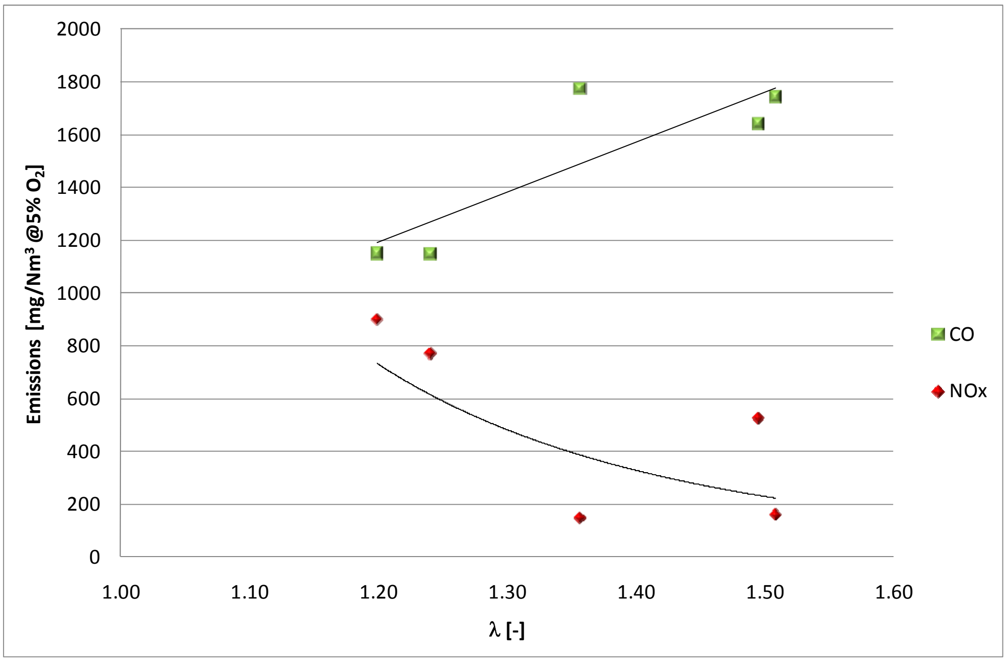

Figure 24 NO_x and CO emissions as function of λ.

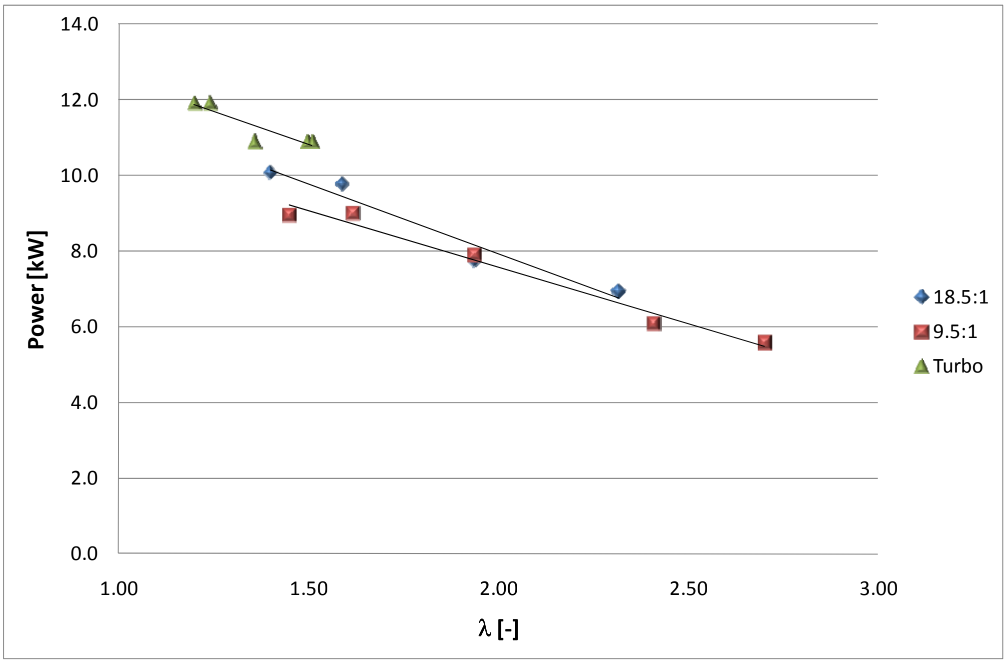

Figure 25 Comparison of the performance for the tree engine configurations.

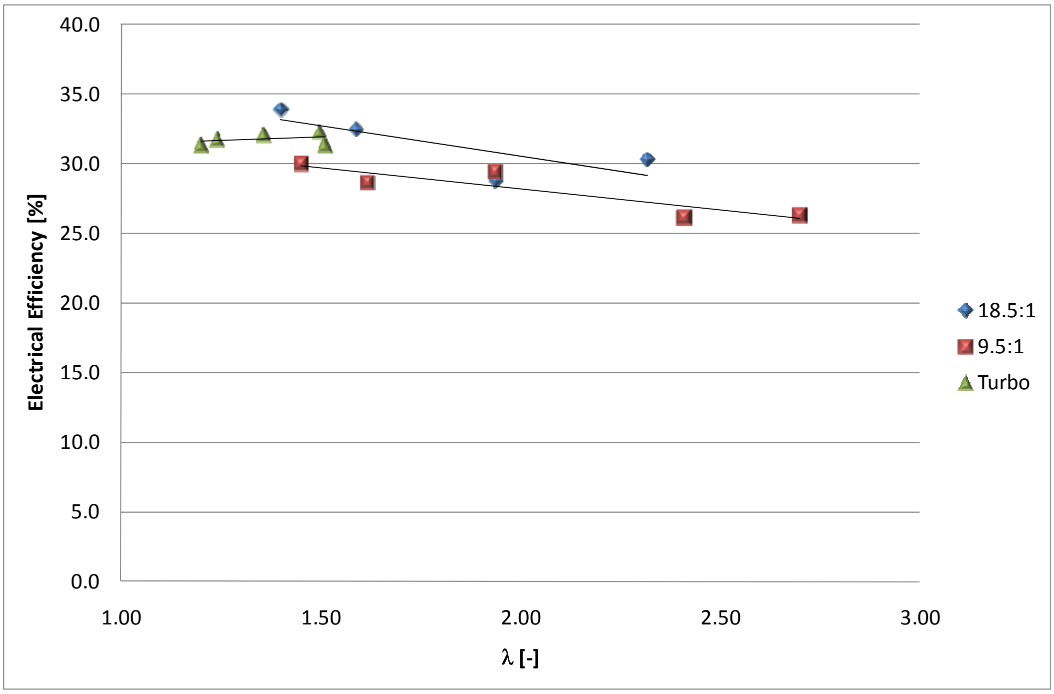

Figure 26 Comparison of the efficiency for the tree engine configurations.

7 Conclusion and Discussion

The project has shown that the conversion of commercial diesel engine gen-sets into high compression spark ignition operation on biomass producer gas is technically possible and economically feasible. An economical assessment showed that for a 200 kW_e gen-set there is a cost difference of approximately 600,000 DKK when the conversion of a diesel gen-set is compared to a conversion of a commercial natural gas engine gen-set. However, it must be noted that it will be difficult to obtain any guaranties for a converted diesel gen-set. On the other hand this is often a problem for natural gas based engine gen-sets also.

In addition to the economical benefits, experimental work has shown that it is possible to operate a converted diesel gen-set with a compression ratio of 18.5:1 on biomass producer gas from the TwoStage Viking gasifier. Two identical small scale converted engine gen-sets were operated on biomass producer gas from thermal gasification of wood. The engines were operated with two different compression ratios, one with a compression ratio for natural gas operation of 9.5:1, and the second with a compression ratio of 18.5:1. It was shown that high compression ratio SI engine operation was possible when operating on this specific biomass producer gas. The experiments showed an increase in the electrical efficiency (from gas to electrical power) from 30 % to 34 % when the compression ratio was increased. This result shows that there is a significant potential for optimisation of engines operating on biomass producer gas. The generator used on the gen-set has an estimated efficiency of 85 % which means that the shaft efficiency of the high compression engine is approximately 40 %. If the engine was coupled with a more efficient generator with an efficiency of 95 %, this would mean that this 15 kW gen-set would have an electrical efficiency of 38 %.

An investigation of the emission characteristics of the high compression engine showed that a significant reduction of the NO_x emission can be achieved by retarding the spark ignition timing. At high load ($\lambda = 1.4$), retarding the ignition timing from 13^o before top dead centre to 7^o reduced NO_x emissions from 1400 to 535 mg/Nm^3.

Final experiments have been conducted with low pressure turbo charging of the high compression engine. The aim was to initiate investigations regarding high compression pressurized engines operating on producer gas. The intake pressure was increased from approximately 500 mmWG under-pressure to approximately 600 mmWG over-pressure. The low pressure charging resulted in a 10 % increase in power output compared to the uncharged high compression engine and 20 % compared to the low compression engine. The efficiency was apparently slightly diminished due to the pressure charging for low λ-values. This may be due to an increase in back-pressure in the exhaust system which was not optimized for pressure charging.

8 Further Work

In continuation of the work carried out in this project it is necessary to investigate the effect of high compression engine operation on engine durability. Experimental measurements of the in-cylinder pressure will make it possible to investigate peak pressure, temperature and stress during high compression combustion and thus it will be possible to predict the durability of the high compression engine concept. Furthermore, the in-cylinder pressure data will make it possible to analyse the combustion process and do further optimisation of the engine concept.

Different modifications of the basic engine design should also be investigated in further experiments based on the experiences from the present project. Focus should be on design parameters including the combustion chamber shape, applied compression ratio, degree of pressure charging, air-fuel ratio and valve timing (Miller cycle). The essential engine parameters to be investigated are heat release, flame speed, optimal ignition timing, heat losses and efficiency. Other important aspects are the necessary after treatment of exhaust gas in the different scenarios in order to fulfil emission requirements.

In addition, the engine concept should also be tested on biogas. From literature it is known that biogas, consisting only of methane and carbon dioxide, has a very high knocking resistance [13] and thus should be an applicable fuel for high compression engine operation.

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